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SPONSORED PROJECT INITIATION

Date: 8/13/79

Project Title: Research in Diesel Engine Noise

Project No: E-16-645

Project Director: W. C. Strahle /J. C. Handley

Sponsor: Detroit Diesel Allison Div. of General Motors Corporation

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Amount: $32,466

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Sponsor Contact Person (s):

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Contractual Matters
(thru OCA)

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Defense Priority Rating: N/A

Assigned to: Aerospace Engineering (School/Laboratory)

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SPONSORED PROJECT TERMINATION

Date: 12/18/80

Project Title: Research in Diesel Engine Noise

Project No: E-16-645

Project Director: W. C. Strahle and J. C. Handley

Sponsor: Detroit Diesel Allison Division of General Motors Corporation

Effective Termination Date: 8/31/80

Clearance of Accounting Charges: 8/31/80

Grant/Contract Closeout Actions Remaining:

X Final Invoice ~~and Closing Documents~~
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_ Final Report of Inventions
_ Govt. Property Inventory & Related Certificate
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CA-4 (1/79)

~~Interim~~ Final Report

RESEARCH IN DIESEL ENGINE NOISE

by

School of Aerospace Engineering

Georgia Institute of Technology

Atlanta, Georgia

for

Detroit Diesel Allison Division of

General Motors

Purchase Order No. B019040

Warren C. Strahle
Principal Investigator

Abstract

Work is reported upon for the first year of a contract between Georgia Tech and Detroit Diesel Allison (DDA) Division of General Motors. Theoretical and experimental work was performed concerning exhaust line noise in turbocharged diesels. Of particular interest was the behavior of the turbine inlet line and the turbocharger turbine on a DDA 8V71T engine. General confirmation was made of the overall noise radiated from the line. The noise was found to be phase locked to the engine processes and not random in nature. Entropy waves were found in the interior of the line but found to be inconsequential in the noise generation processess. A technique was investigated to determine the turbine admittance. Exploratory runs determined the feasibility of the method and an important result was found that the turbine speed is essentially constant under the action of the nearly periodic impulses.

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Introduction

There has been relatively little work done on noise generation by exhaust lines in diesel engines, especially for the turbocharged case. The turbocharger introduces a component which at once both aids and detracts from noise generation in the exhaust line. For one, it is a barrier to the pressure waves coming from the exhaust ports and so holds up the pressure level in the line from the ports to the turbine. Secondly, since the turbine extracts work from the exhaust gases, it is supposed that the exhaust line downstream of the turbine would have diminished excitation, as opposed to the non-turbocharged case. The turbine itself can cause noise due to the interaction of the pulsatile flow with the turbine processes.

To investigate the role of the turbocharger in exhaust line noise Georgia Tech embarked on a program sponsored by DDA in September, 1979. The program used an 8V71T engine supplied by DDA, together with transmission and dynamometer. Initial goals were to a) confirm or deny prior DDA results on the level of exterior noise generated by the thin walled lines connecting the exhaust ports to the turbine and b) develop methods to investigate the role of the turbocharger in noise production or suppression. The level of effort was approximately 1/2 man year.

While substantial progress was made toward the goals, the program was not without its problems, mostly due to personnel. A graduate student assigned to the project dropped out of school in mid-year, and the post-doctoral fellow who did most of the work did not get well under way until 3 months into the project. Moreover, near the end of the year an accident destroyed the original turbocharger and caused some delay.

Two program reviews were held with DDA personnel and some of the results presented here have been orally communicated. The results which are totally new are those having to do with the measurement of the isentropic and non-isentropic turbine admittance.

Apparatus and Instrumentation

The engine used was a DDA 8V71T engine, supplied by DDA together with water brake dynamometer and transmission. For turbocharger turbine intake line interior measurements, 4 access ports for instrumentation were constructed as shown in Figs. 1 and 2. These ports were to receive both pressure and temperature instrumentation. Either low impedance Photocon or high impedance AVL, all water cooled, pressure transducers could be accommodated. Thermocouples used were .001 in diameter chromel-alumel thermocouples which had a time constant of about .05 sec and were compensated by the method of Ref. (1).

For the study on the turbine behavior a set of access ports were constructed as shown in Fig. 3. For external noise measurements 1/2 inch B & K microphones were used. For turbine wheel speed measurements an instrumented turbocharger with a magnetic pickup were used with a frequency to voltage converter. Engine speed and torque were monitored.

All signals were recorded on an FM tape recorder for later Fourier analysis. Data reduction was performed in the Fourier analysis equipment.

Theory of Turbine Behavior

Two theoretical efforts were conducted on this program, both related to deducing the turbine acoustic impedance or admittance. The specific acoustic admittance relation which must hold at the entrance plane to the turbine for plane waves may be stated as

$$\frac{u'_\omega}{\bar{c}} - \frac{A}{M\gamma}\frac{p_\omega}{\bar{p}} - B\frac{s_\omega}{c_p} = 0 \tag{1}$$

where the admittance coefficients are A and B. For isentropic waves, the significance of A is that it gives the relation between pressure and velocity at the turbine entrance plane. In fact, it may be shown[2] that the acoustic intensity traveling into the turbine is given by (for a single frequency tone)

$$I = \frac{<p'^2>}{\bar{\rho}\,\bar{c}}\left\{A(1+M^2) + A^2M + M\right\} \tag{2}$$

Consequently, for a given pressure fluctuation level, the larger A is the larger is the acoustic intensity travel into the turbine. The expectation is that this would be a nice feature of the turbine, since it would alleviate the pressure fluctuation in the turbine inlet line in favor of acoustic energy going toward the tail pipe muffler.

The significance of B is that entropy waves incident on the turbine will induce isentropic fluctuations in u and p. The turbine has a large pressure gradient in the interior flow which is known to yield a non zero B.[3]

One of the theoretical items of this work was the construction of a one dimensional theory of the acoustics of the turbine inlet line. The purpose was to express the unknown turbine admittance in terms of the measurable pressures at the instrumentation ports. However, it was found experimentally that the error in the measured admittance was too large by this technique and the method was abandoned. Consequently, this theory will not be presented here.

An estimate of the turbine behavior can be constructed utilizing some simplifying assumptions. Supplied was a map of the mass flow characteristic of the turbine. This is reproduced in Fig. 4. If it is assumed that the turbine responds quasi-statically (i.e. follows this mass flow characteristic) even under oscillatory conditions, the admittance coefficients can be calculated. Crucial to this assumption is that the wheel speed would also adjust instantaneously, which it obviously cannot do. However, lines of constant wheel speed were also provided and are sufficiently near tangency to Fig. 4 that it is believed little error is made. Physically it would be expected that the gasdynamic processes would behave quasi-statically if a flow time through the turbine is short compared with a cycle time for the frequency under consideration. Therefore, the assumption would appear to fail somewhere in the 100-1000 Hz range, but should be fairly good (except for the wheel speed problem) below 100 Hz.

A second assumption is needed regarding p_{2_t}. Experimentally, there is a reduction of the fluctuation in $p_2 \approx p_{2_t}$ as compared with the fluctuation in p_1. Consequently, it was assumed $p_{2_t} = 0$. Linearizing the mass flow curve for small fluctuations and using perfect gas relations together with the above assumptions the following relations emerge for the admittance

$$A = \frac{-M\left[1 - (\gamma-1)\ \zeta - \xi\gamma\right]}{1 - (\gamma-1)\ M^2\ (\zeta + \frac{\gamma\xi}{\gamma-1})}$$

$$B = 1/2 \tag{3}$$

where

$$\zeta = \frac{1}{\eta}\left[\frac{\gamma\xi}{\gamma-1}(1-\eta) - 1/2\right]$$

$$\xi = \left(\frac{g'}{\bar{g}}\ \frac{\overline{p_1}_t}{\bar{p}_{2_t}} + 1\right)$$

$$\eta = 1 + \frac{\gamma-1}{2} M^2$$

The relation for A is plotted in Fig. 5, where the M = 1 limit is shown as $(\gamma - 1)/2$. There is seen a tremendous variation in behavior as the pressure ratio is changed. However, it is noted that analytically the formula becomes heavily sensitive to the slope of the mass flow characteristic as the pressure ratios becomes high. Consequently, the high pressure ratio results are shown as a dashed line.

The Mach number, M, in the above has been the Mach number at the minimum area point in the nozzle. However, measurements are going to be made upstream of the nozzle and it is desirable to correct the A and B numbers for wave motion in the converging nozzle. This was done using Candel's[3] one dimensional theory. This requires numerical integration of three coupled ordinary differential equations of complex numbers. This was performed but the details will be omitted. A representative calculation was made for a mid-load, mid-speed condition, and the results are shown in Fig. 5. The dimensionless frequency upon which the coefficients depend is $f = \omega l/u_o$ which corresponds to a physical frequency of 192 Hz for the parameters used in this computation. The major point to notice is that B falls rapidly with frequency whereas the A variation is rather small. Therefore, above 400 Hz, say, there should be little generation of sound if entropy waves exist in the turbine intake line. It also appears that a highly loaded engine will have a low admittance in general so it will be highly reflective to accident waves.

This last observation leads to one recommendation of this work. A look should be made at a turbine matched to give the required output with low turbine load (p_{1_t}/p_{2_t}). That is, use an oversized turbine. The calculations suggest that significantly more acoustic energy would be passed downstream, toward the muffler.

Experimental Results

Turbine Intake Line

Three basic runs of the engine were made to deduce properties of the interior behavior of the line and the exterior noise characteristics. All runs were made at the mid-load mid-speed conditions of 1800 RPM, 150 HP. The first run was exploratory in nature to check out instrumentation and to gain familiarity with the two-microphone technique of acoustic intensity measurement.[(4)] The second run was made to deduce noise characteristics of the turbine inlet line in the presence of overall engine noise contamination and to investigate the interior pressure and temperature fluctuation data. The third run was made with all engine components, except the turbine inlet line, wrapped with fiberglass. Under this last condition the only audible noise signal came from the turbine inlet line and the turbocharger intake.

Consider first the interior measurements. A typical pressure-time trace, which has been averaged over 100 cycles triggered at a fixed crank angle, is shown in Fig. 7. A fair amount of randomness occurs from cycle to cycle. This is illustrated by subtracting the mean diagram of Fig. 7 from the last of the 100 samples and the result is shown in Fig. 8. Apparent is basically higher frequency random behavior. This is further illustrated by the pressure spectra of Fig. 9. The upper curve is the ensemble averaged spectrum, containing the total signal. The lower curve is the magnitude of the square of the Fourier transform of the ensemble averaged pressure-time diagram where the random behavior has been removed by averaging.

In Fig. 9 are seen the harmonics of the fundamental firing frequency of 30 Hz. Moreover, a) these peaks contain the majority of the mean square pressure and b) random behavior does not take over until above about 500 Hz. These observations are a more precise version of what was gleaned from Figs. 7 and 8, viewing the pressure time trace. The most striking feature of the spectra is the irregularity in the peak heights; there appears no order to them and the fourth harmonic is the dominant one as opposed to the first, which may have been expected to be the largest. This complex behavior is probably caused by two factors. First, the turbine inlet line flow is being excited by a space distributed set of exhaust ports and the lines possess a complex acoustic response as a function of frequency. Secondly, the period between firings (and exhaust) is not uniform from cylinder to cylinder because of the engine design.

Consider now the temperature fluctuations, where a typical spectrum is shown in Fig. 10. The compensation of the thermocouple is valid out to about 300 Hz, above which it is evident the signal has hit the electronic noise floor. Below this frequency the temperature fluctuations are dominantly periodic. The important point is that there is not a one to one correspondence between the relative peak heights here as compared with the peak heights of the pressure. This indicates non-isentropic behavior due to either a) nonlinearities in the acoustics or b) the turbulence and shock wave problem of the exhaust jets entering the turbine inlet line. In any event, there will be entropy waves travelling toward the turbine and the full admittance relation of Eq. (1) may have to be considered.

One final result from the interior measurements has a bearing on the turbine admittance. Figure 11 shows the spectrum of the turbine outlet pressure as compared with that at the turbine inlet. It is clear that while there is a drop in the fluctuation level across the turbine, as expected, the outlet line fluctuation is not negligible, as was assumed in the previous section. Future work should remove this analytical restriction. The physical expectation is that this phenomenon would mean the turbine is less of a barrier (higher admittance) to acoustic energy than calculated previously. This will be seen to indeed be the case from the admittance measurements presented later.

Consider now some representative exterior noise measurements made with both the wrapped and unwrapped engine and using the two microphone technique for intensity measurements[4]. The "near field" measurements are made two inches radially outward from the turbine line. For a wrapped engine a survey along the line is shown in Fig. 12. The points at (2) and (3) are heavily contaminated by compressor inlet noise, since the compressor intake was not silenced. In any event a coarse estimate of the sound power coming from the line is constructed by assuming the intensity is the same as measured if one were to move circumferentially around the line. The result is a sound power of 9.22 mW. This number is of the same order of magnitude as that measured by DDA by a more precise method. This is compared with a single point far field measurement estimate for the unwrapped engine of 203 mW. A typical spectrum of the acoustic intensity is shown for position (4) of Fig. 12 in Fig. 13. Evident at low frequency are the peaks at multiples of the firing frequency with a general decay of about 10 dB/decade above about 300 Hz.

The coherence between the acoustic pressure outside and inside the line is shown in Fig. 14. for a wrapped engine. This is as expected where the dominant periodic peaks observed in both interior and exterior spectra also show high coherence. While this does not prove a causal relation between the interior and exterior processes, at least a zero or low coherence would have shown no causality. The latter occurence could not appear because both interior and exterior spectra show the periodic peaks.

Turbine Behavior

It was mentioned earlier that the original technique envisioned for determination of the turbine admittance turned out to be unsatisfactory. Although an oral presentation of those results was made, they are not presented here. Late in the program it was decided to attempt a direct measurement of the admittance relation. Some progress was made, but an accident with ingestion of metal into the compressor and subsequent unbalance and clashing of the turbine with the housing caused delays and prevented the desired degree of progress.

It was decided to attempt to use the two microphone technique on the interior flow at the turbine entrance, in order to measure the velocity in Eq. (1). Using perfect gas relationships, a relation between the entropy, temperature and pressure transform is

$$\frac{s_\omega}{c_p} = \frac{t_\omega}{\bar{t}} - \frac{\gamma - 1}{\gamma}\,\frac{p_\omega}{\bar{p}} \qquad (4)$$

Placing Eq. (4) into Eq. (1) and constructing cross spectra by first multiplying by $t_{\omega}^{*}/\bar{t}$ and then by $p_{\omega}^{*}/\bar{p}$, two equations are generated in the unknowns A and B.

$$\frac{S_{ut}}{\bar{u}\bar{t}} - \left(\frac{A}{\gamma M} - \frac{\gamma-1}{\gamma} B\right) \frac{S_{pt}}{\bar{p}\bar{t}} - B \frac{S_{tt}}{\bar{t}^2} = 0$$

$$\frac{S_{up}}{\bar{u}\bar{p}} - \left(\frac{A}{\gamma M} - \frac{\gamma-1}{\gamma} B\right) \frac{S_{pp}}{\bar{p}^2} - B \frac{S_{tp}}{\bar{p}\bar{t}} = 0 \tag{5}$$

If B = 0, as, recall, it is expected to be at sufficiently high frequency, any one of the two equations may be used to calculate A.

Six runs were made under conditions shown in Table 1. In the data reduction process it was found that highly erratic results were produced if data were used at frequencies removed from harmonics of the firing frequency. It will be recalled that except at these harmonics the spectra indicate random behavior and so do not belong to the class of propagational acoustic flows for which Eq. (1) is valid. Consequently, only data at the harmonic frequencies were finally used.

Table 1

Run Conditions for Admittance Determination

Engine Speed	Power	Wheel Speed
RPM	HP	RPM
1800	0	34020
1800	125	52140
1800	150	57960
1800	175	57480
2100	150	58500
1500	150	49500

An extremely important side issue was the dynamic behavior of the wheel speed. Consequently, continuous recording of the speed was also made. Covering this issue first, consider the spectrum of the wheel speed in Fig. 15. The separation of the two spectra under the two types of averaging clearly show the behavior to be random, with no harmonics of the engine speed appearing. More revealing, and not shown in the figure, is that this spectrum is at least 50 dB below the d.c. reading. Consequently, for all practical purposes the wheel speed is a constant at fixed load and speed. This has important implications in future analytical efforts on prediction of the turbine admittance.

Turning to the admittance measurements, it was found upon data reduction that the thermocouple had failed just after the no-load run at 1800 RPM. Consequently, the full data reduction was only accomplished for that run, but this did not mean an invalidation of those runs, as will be seen. Figure 16 shows the magnitude of B, confirming the expectation of dropping to zero after only a few hundred Hz. However, the magnitude near zero frequency is far too high, and it is believed that a fundamental problem exists with the measurement method that will require more development. It is believed that severe calibration problems exist when the two microphone technique is used for a hot, interior measurement. Especially at low frequency, where the wavelengths are long, significant error can arise when all calibration is done at room temperature conditions.

Looking at Fig. 17, where $|A|$ is displayed, the same suspicion is present. While the theory of Fig. 6 is not touted as the ultimate in accuracy, it was believed that the magnitude and trends with frequency should be correct. The experiment of Fig. 17 shows magnitudes far too high, especially at low frequency.

Using the fact that B does, however, fall to zero rapidly, the data reduced with B set identically zero are shown over an expanded frequency range in Fig. 18. The general constancy in magnitude is believed, but it again appears that a systematic error in overestimating the velocity is being made.

For completeness, although it shows the same problem, a run at load is presented in Fig. 19. The basic conclusion is that more work is necessary, especially with the velocity measurement, to produce reliable turbine admittance values.

Conclusions

1. There is general agreement between this laboratory and DDA as to the level of noise radiation from the turbocharger turbine inlet lines.

2. The inlet lines have interior wave motions consisting of isentropic and entropy waves. The oscillations are dominantly periodic, though complex.

3. The entropy waves should not generate any turbine excitation beyond a few hundred Hz, and may therefore be largely ignored from the noise generation standpoint.

4. The determination of the turbine admittance remains of interest to the noise generation process; this determination was not accomplished here but remains a goal for future work.

5. Based upon two experimental results here, those of the downstream pressure fluctuation in the exhaust line and the constancy of the wheel speed, a new analytical model of the turbine admittance should be constructed.

6. An attempt should be made to force a low admittance turbine by using an oversized turbine; this should lower the turbine inlet line noise.

7. The two microphone technique <u>may</u> not be suitable for an interior velocity measurement. Other velocity measurement tools should be investigated.

Nomenclature

A	isentropic admittance coefficient
B	non isentropic admittance coefficient
c	speed of sound
c_p	specific heat at constant pressure
g	mass flow function of p_{2_t}/p_{1_t}
ℓ	length of turbine inlet
I	acoustic intensity
M	Mach number
p_ω	pressure fluctuation transform
s_ω	entropy fluctuation transform
S_{ij}	cross spectral density between signal i and signal j
t	temperature
t_ω	temperature fluctuation transform
u	axial velocity
u_ω	plane wave velocity fluctuation transform
γ	ratio of specific heats
ρ	density
ω	frequency

Superscripts

$'$	fluctuating quantity or derivative
$-$	mean quantity

Subscripts

t	total
1	upstream of turbine
2	downstream of turbine
v	temperature spectrum
p	pressure spectrum
w	wheel speed spectrum

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1. Strahle, W. C. and Muthukrishnan, M., "Thermocouple Time Constant Measurement by Cross Power Spectra," AIAA J., Vol. 14, pp 1642-43 (1976).

2. Goldstein, M. E., Aeroacoustics, McGraw Hill, New York, p. 42 (1976).

3. Candel, S. M., "Analytical Studies of Some Acoustic Problems of Jet Engines," Ph.D. Dissertation, Calif. Inst. of Tech. (1972).

4. Chung, J. Y., "Cross-spectral Method of Measuring Acoustic Intensity without Error Caused by Instrument Phase Mismatch," J. Acoust. Soc. Am., Vol. 64, pp 1613-1615 (1978).

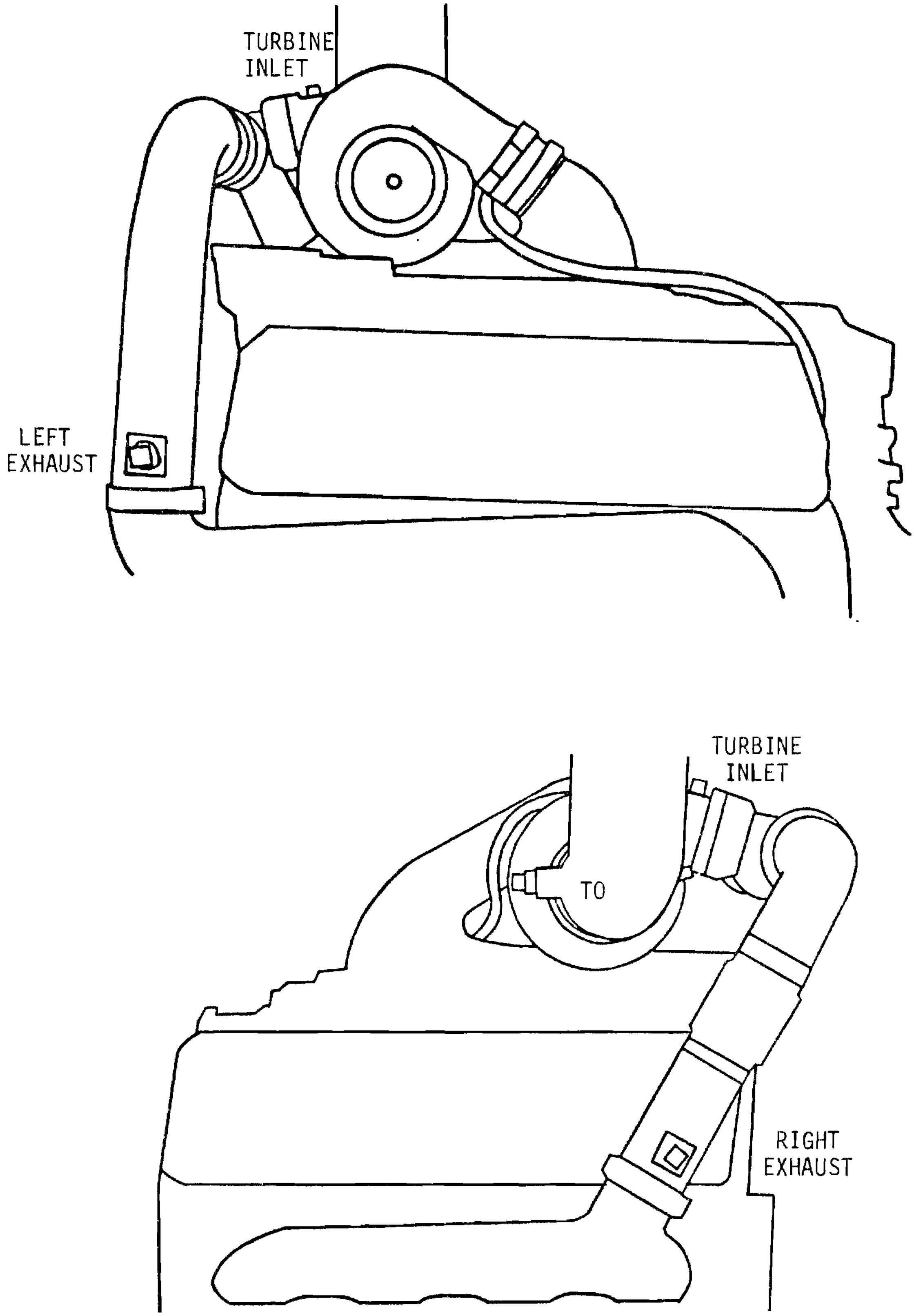


Figure 1. Instrumentation ports - side views

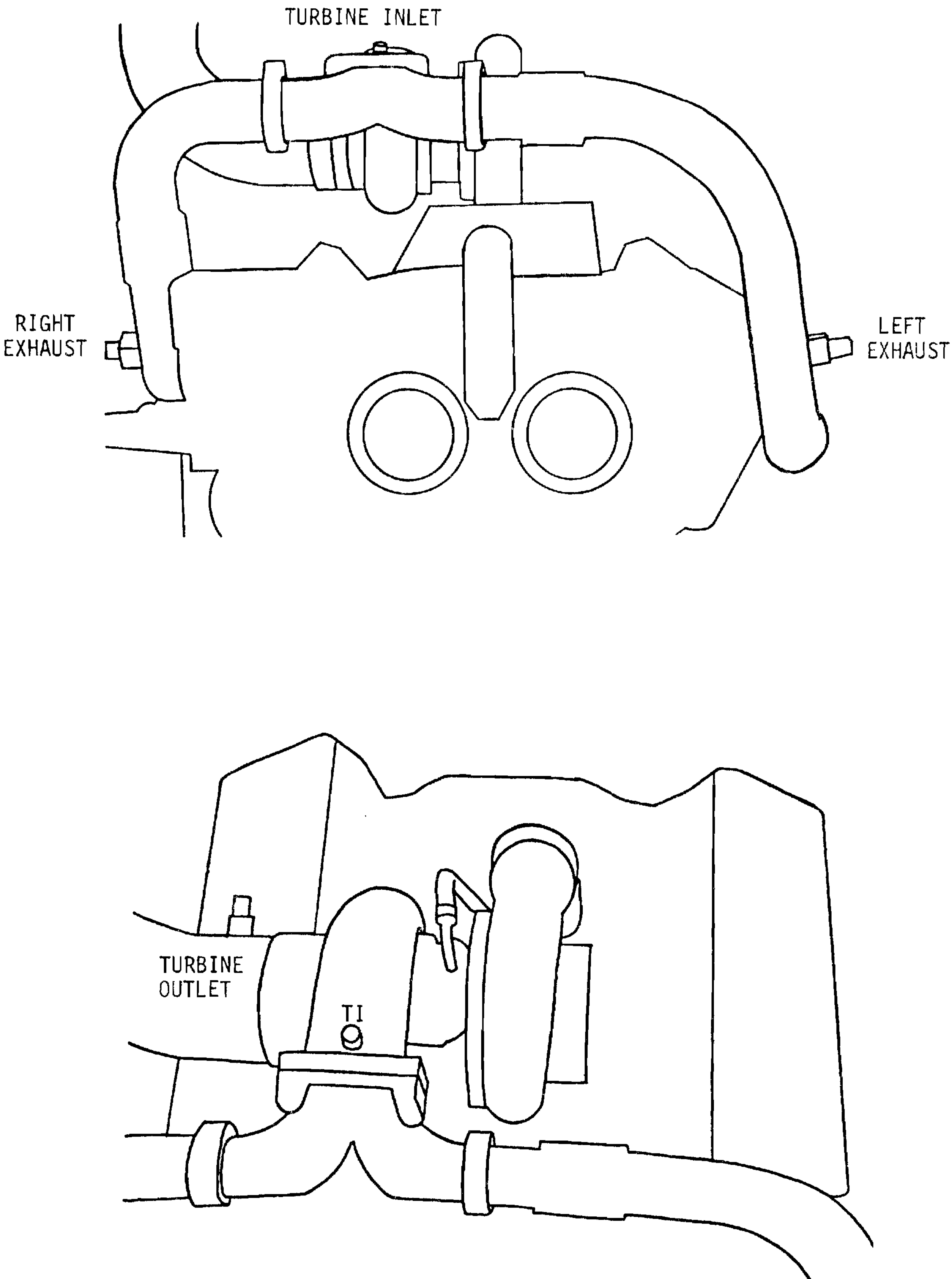


Figure 2. Instrumentation ports - top and front views

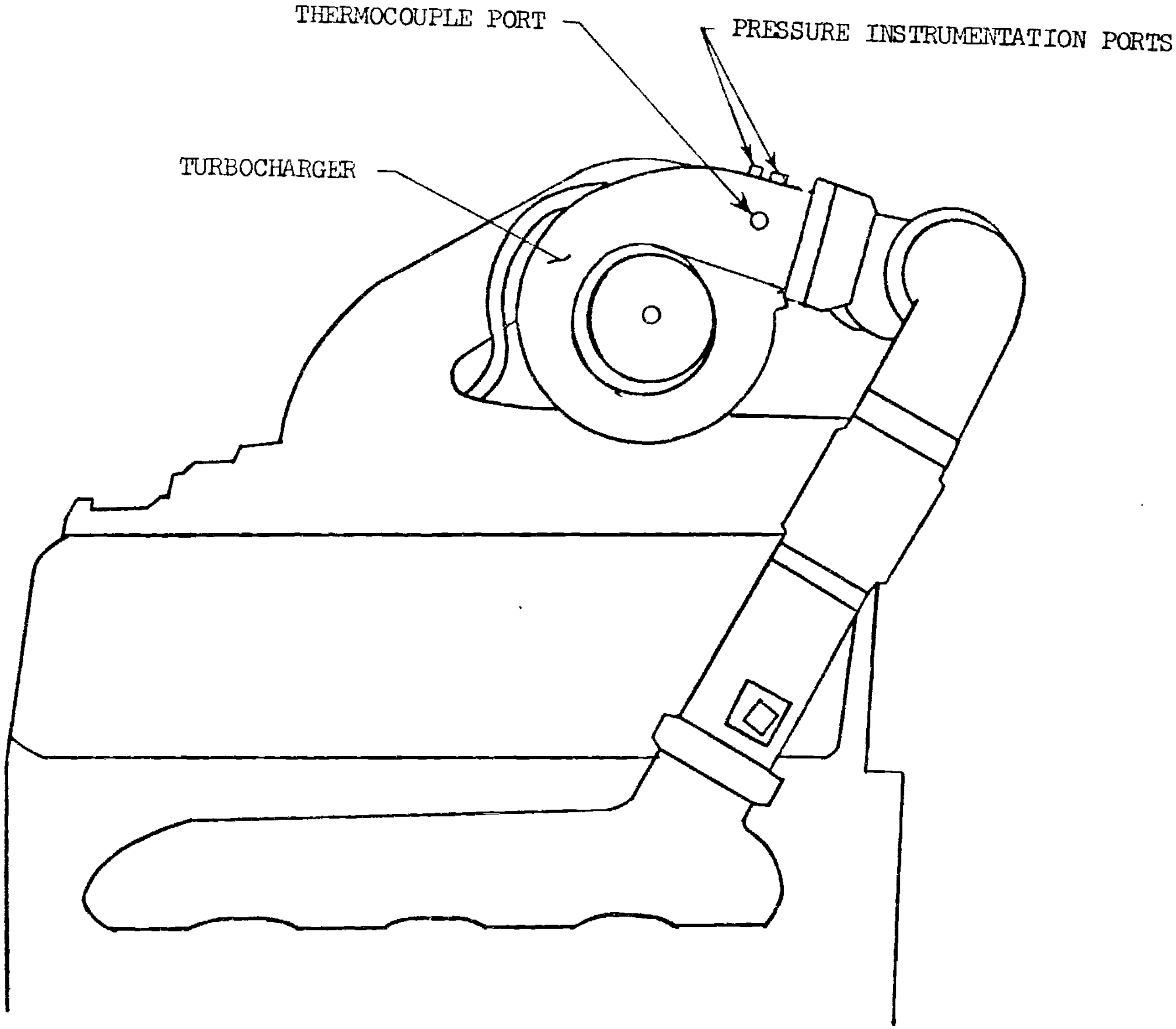


Figure 3. Turbine instrumentation ports

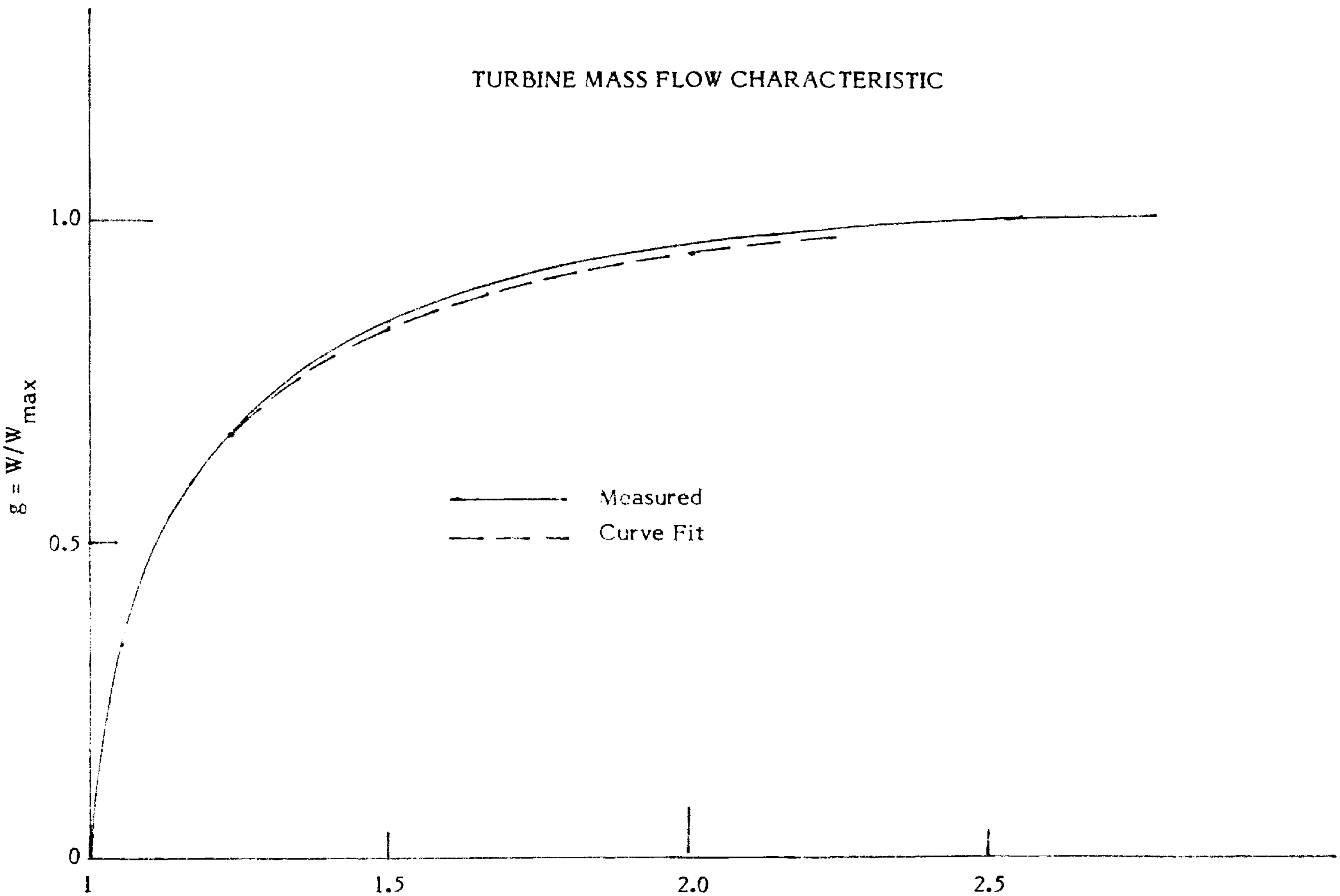


Figure 4. Turbine mass flow characteristic

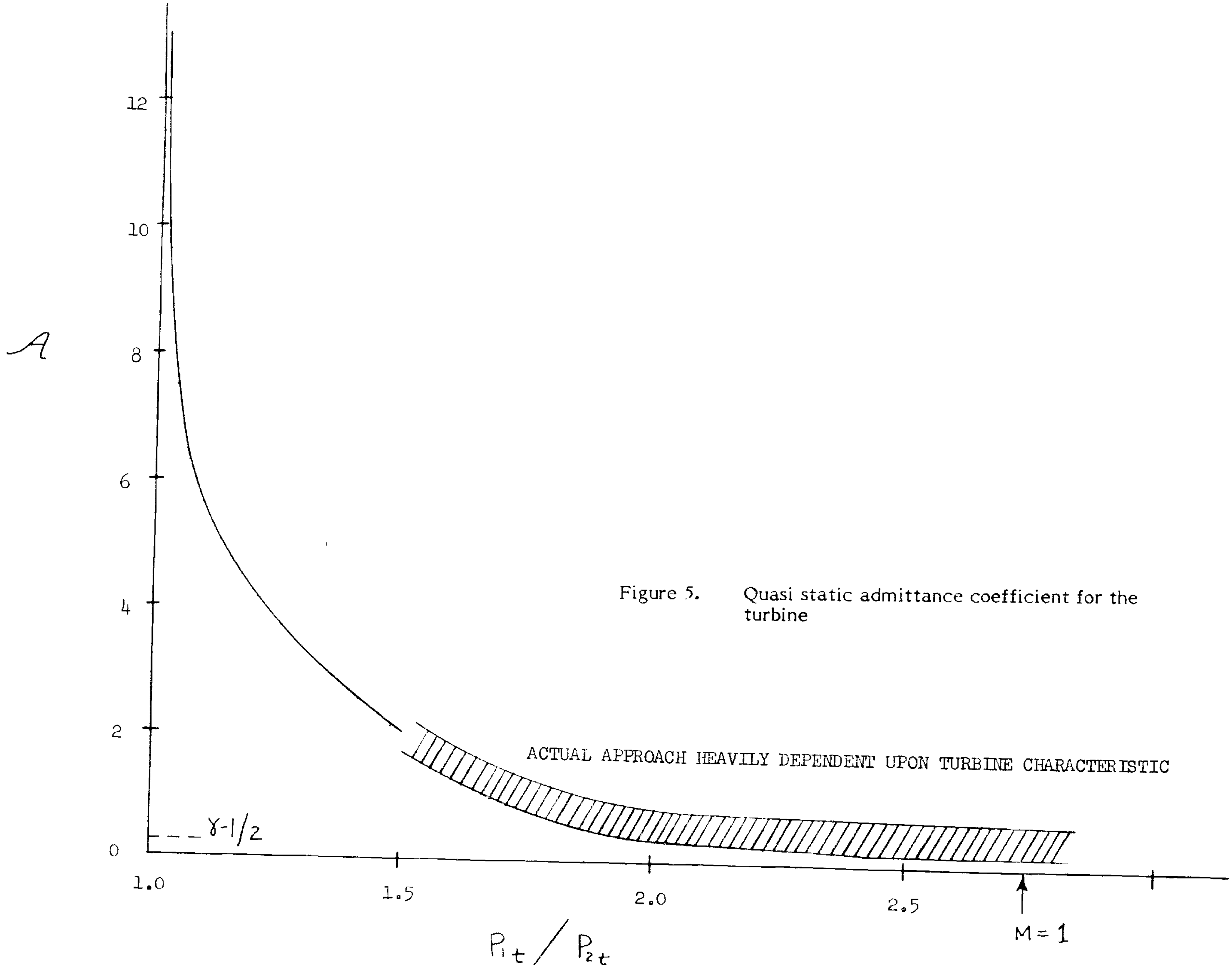


Figure 5. Quasi static admittance coefficient for the turbine

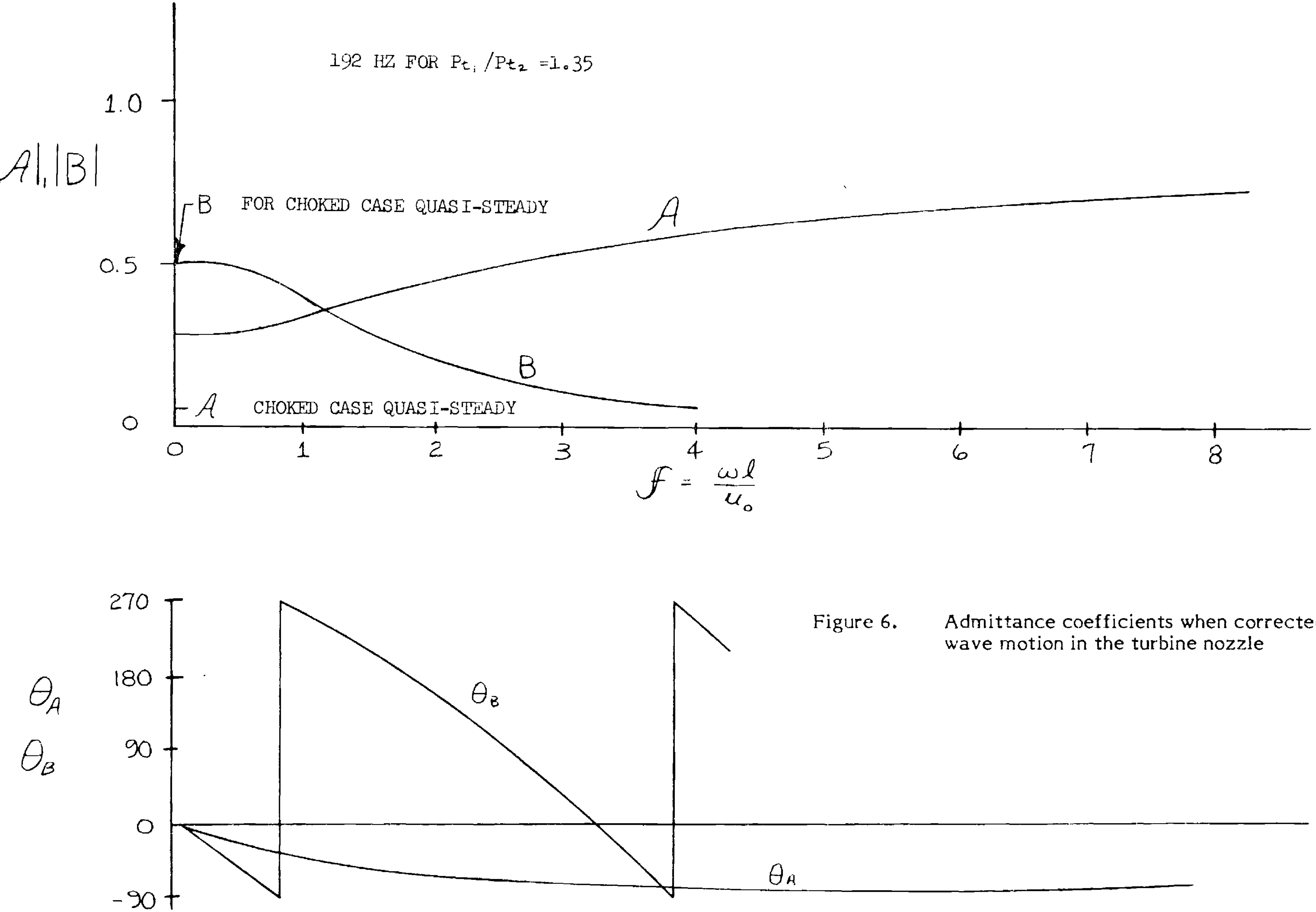


Figure 6. Admittance coefficients when corrected for wave motion in the turbine nozzle

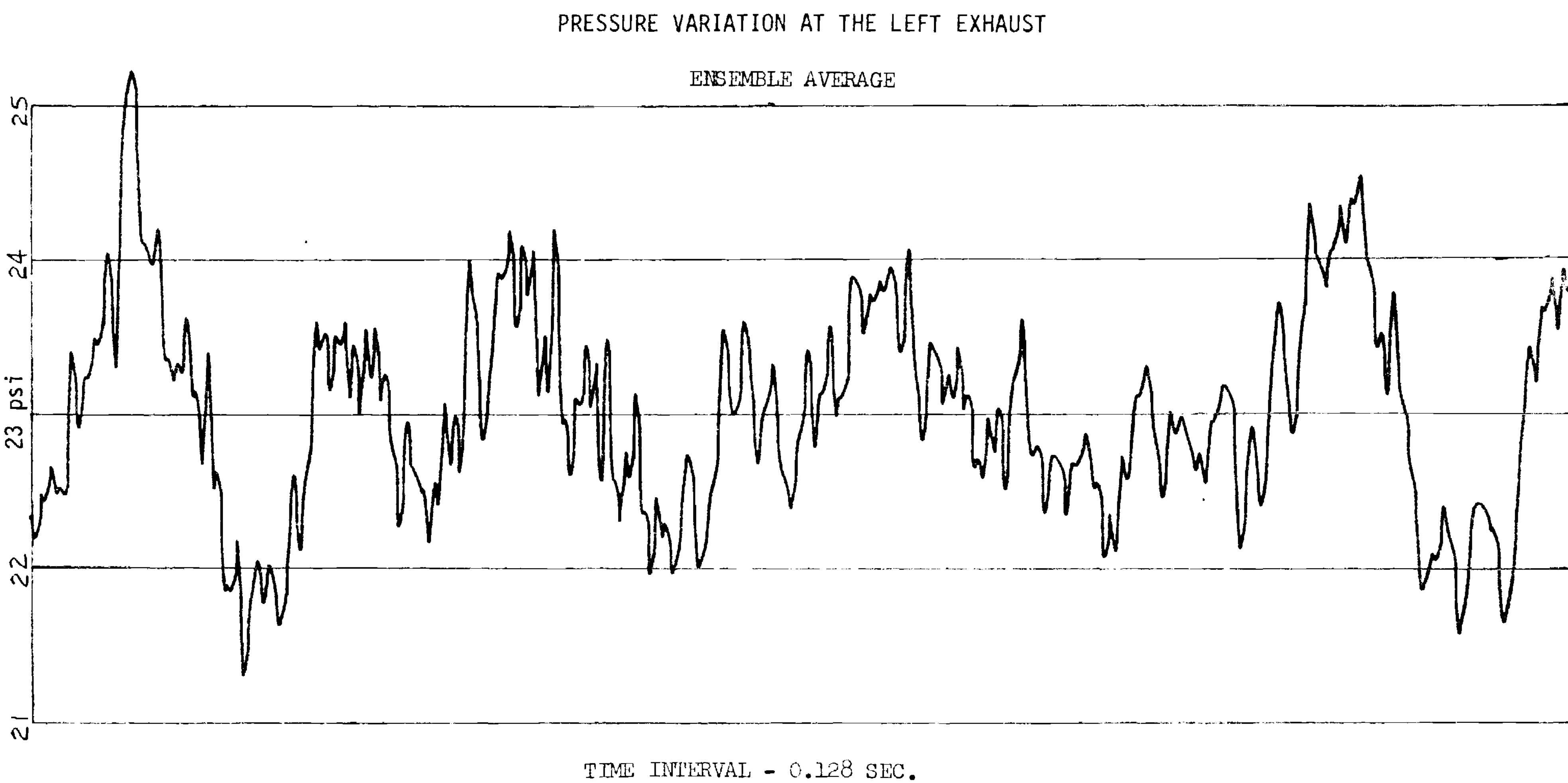
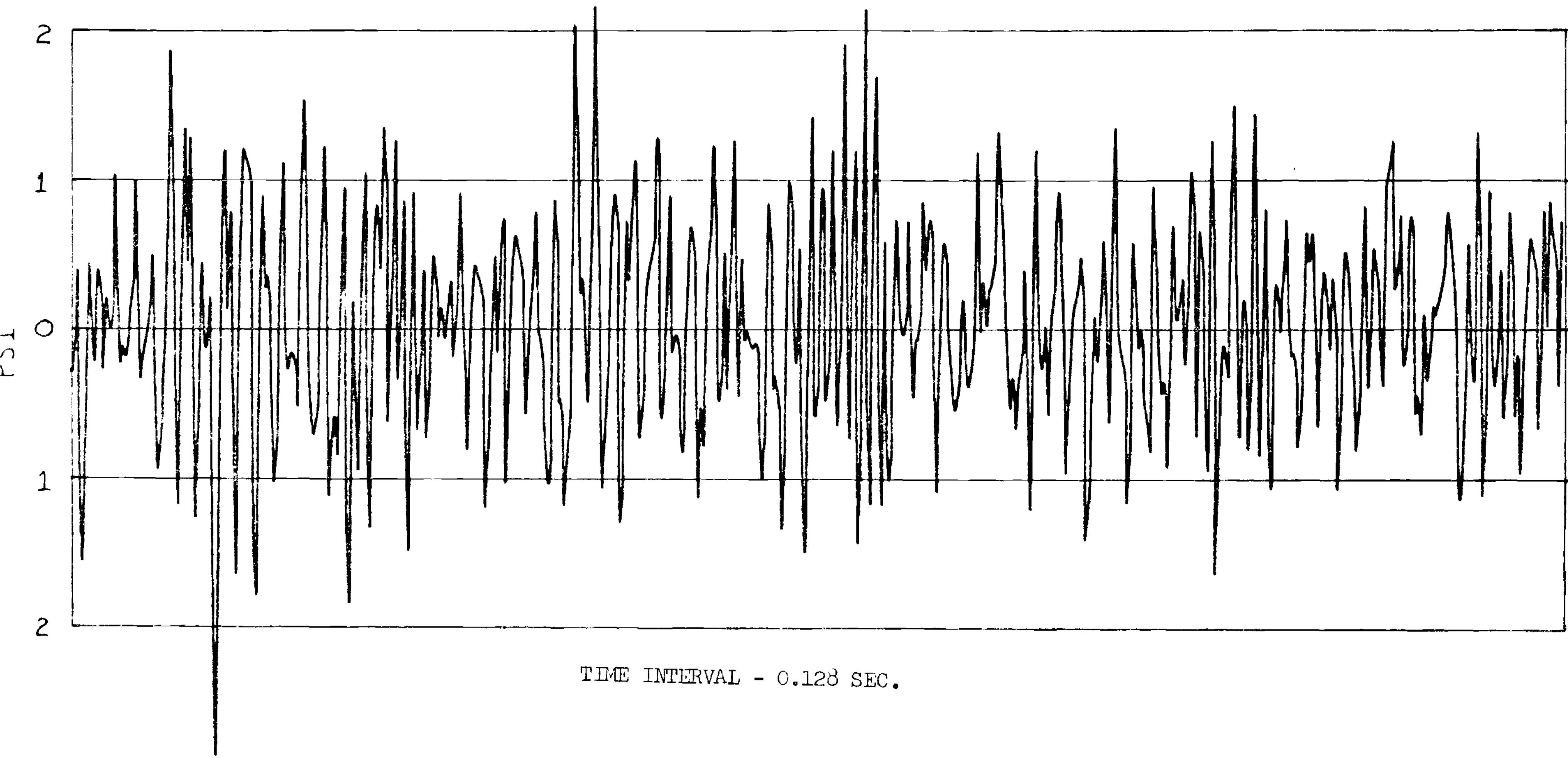


Figure 7. Pressure-time trace measured at the left exhaust

PRESSURE VARIATION AT THE LEFT EXHAUST

RANDOM FLUCTUATIONS



TIME INTERVAL - 0.128 SEC.

Figure 8. Subtraction of Figure 7 from the last of the 100 samples used. Random behavior of the pressure time diagram

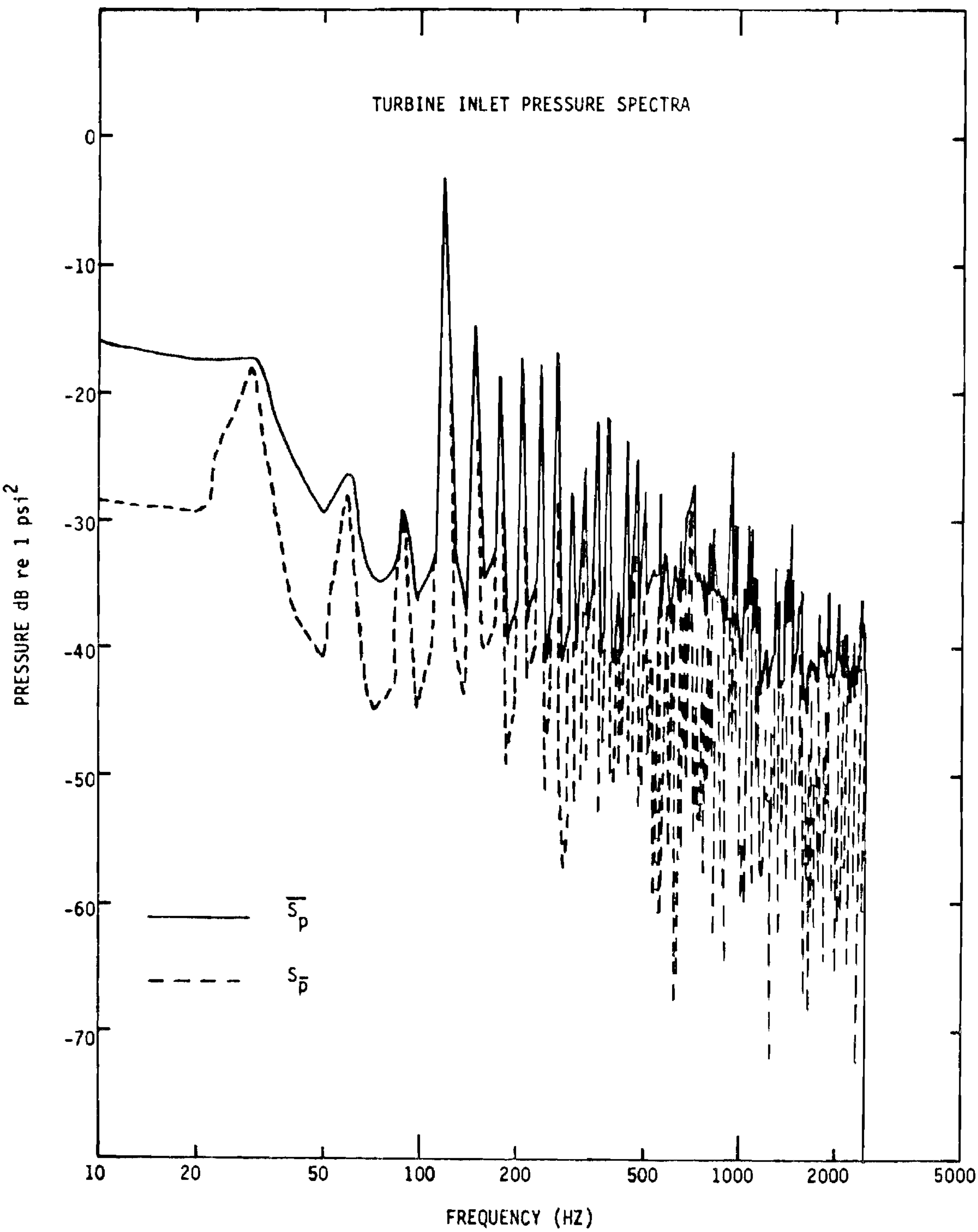


Figure 9. Ensemble averaged pressure spectra and spectra of the ensemble average pressure at the turbine inlet

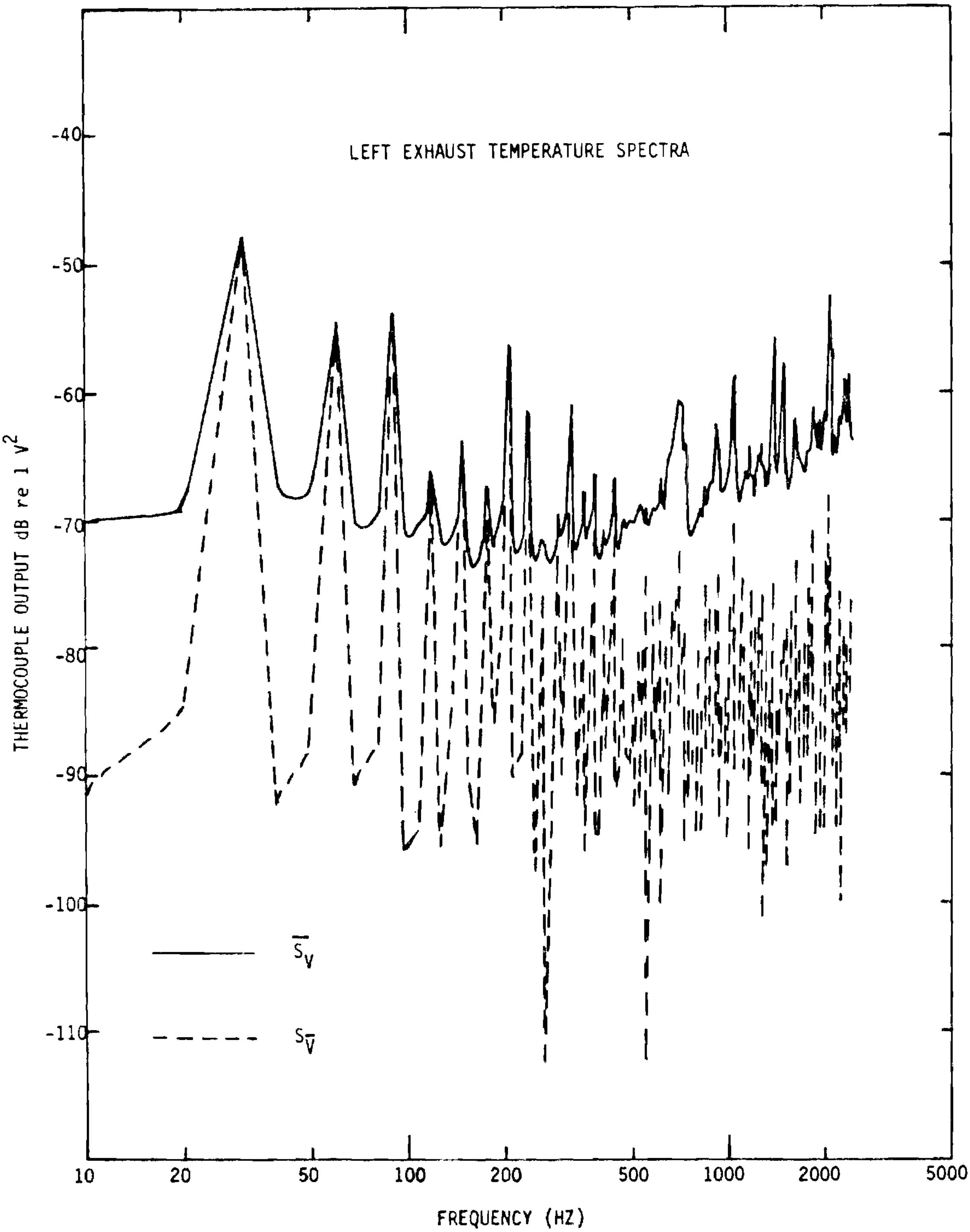


Figure 10. Temperature spectra at the left exhaust

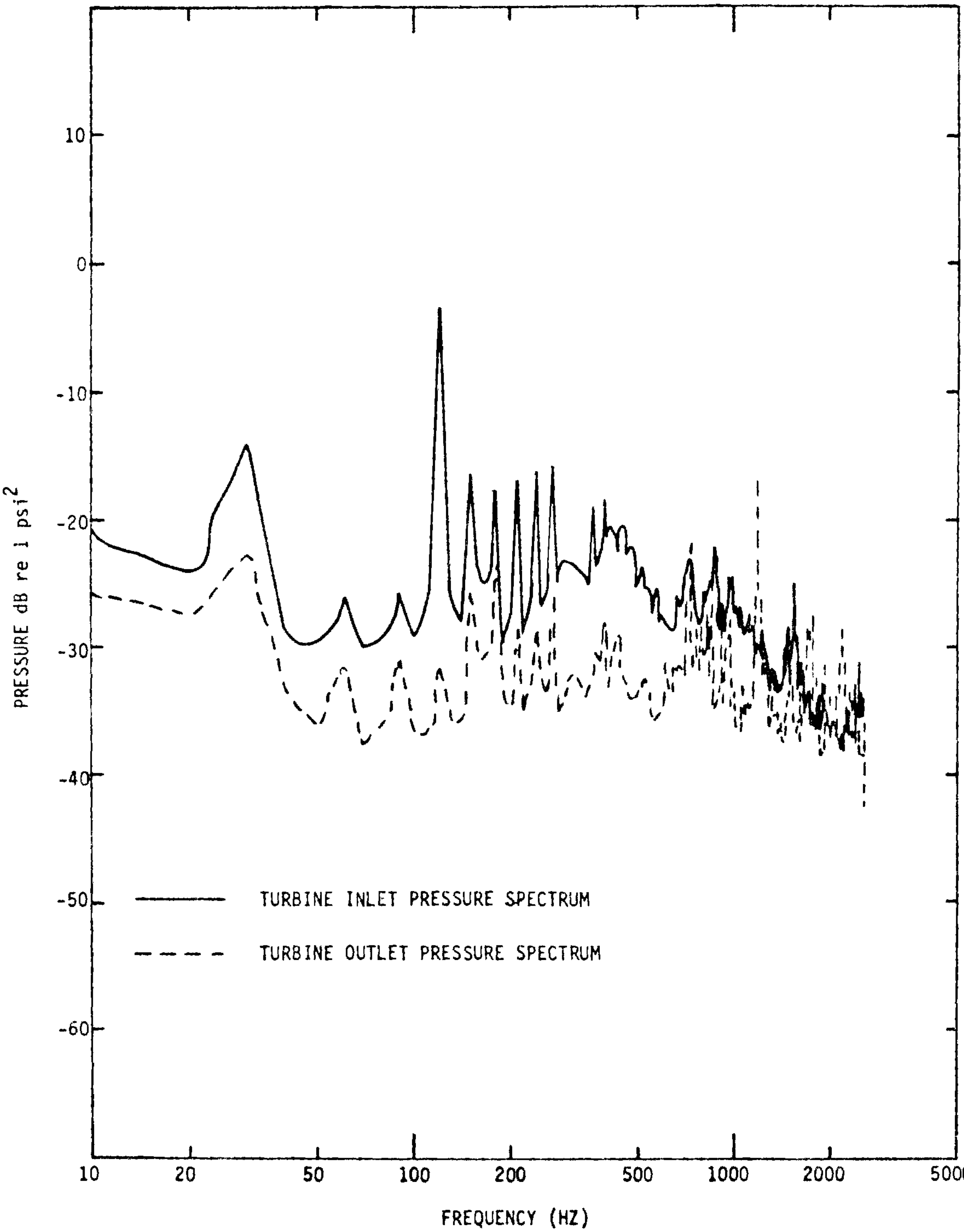
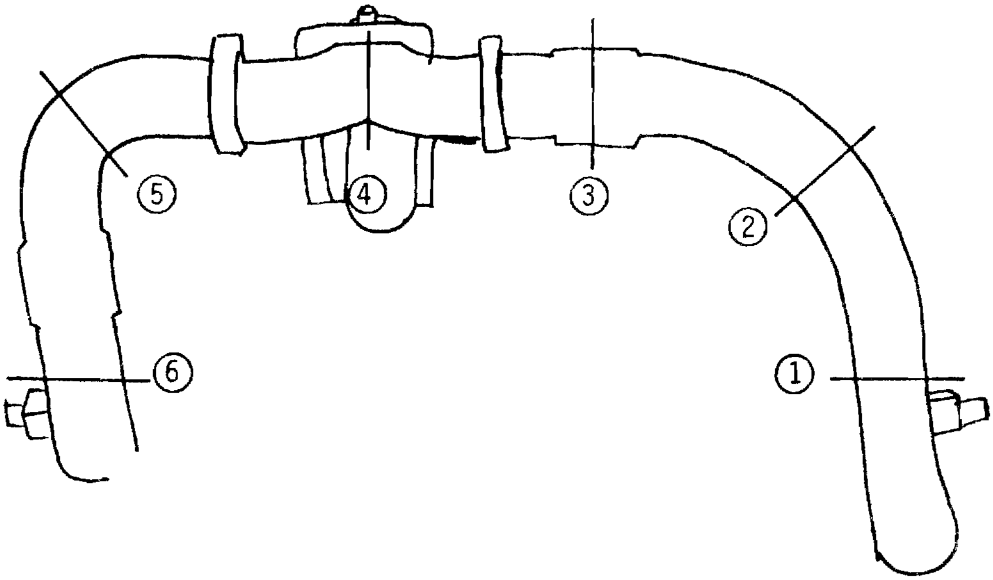


Figure 11. Comparison of the pressure spectra at the inlet and outlet of the turbine



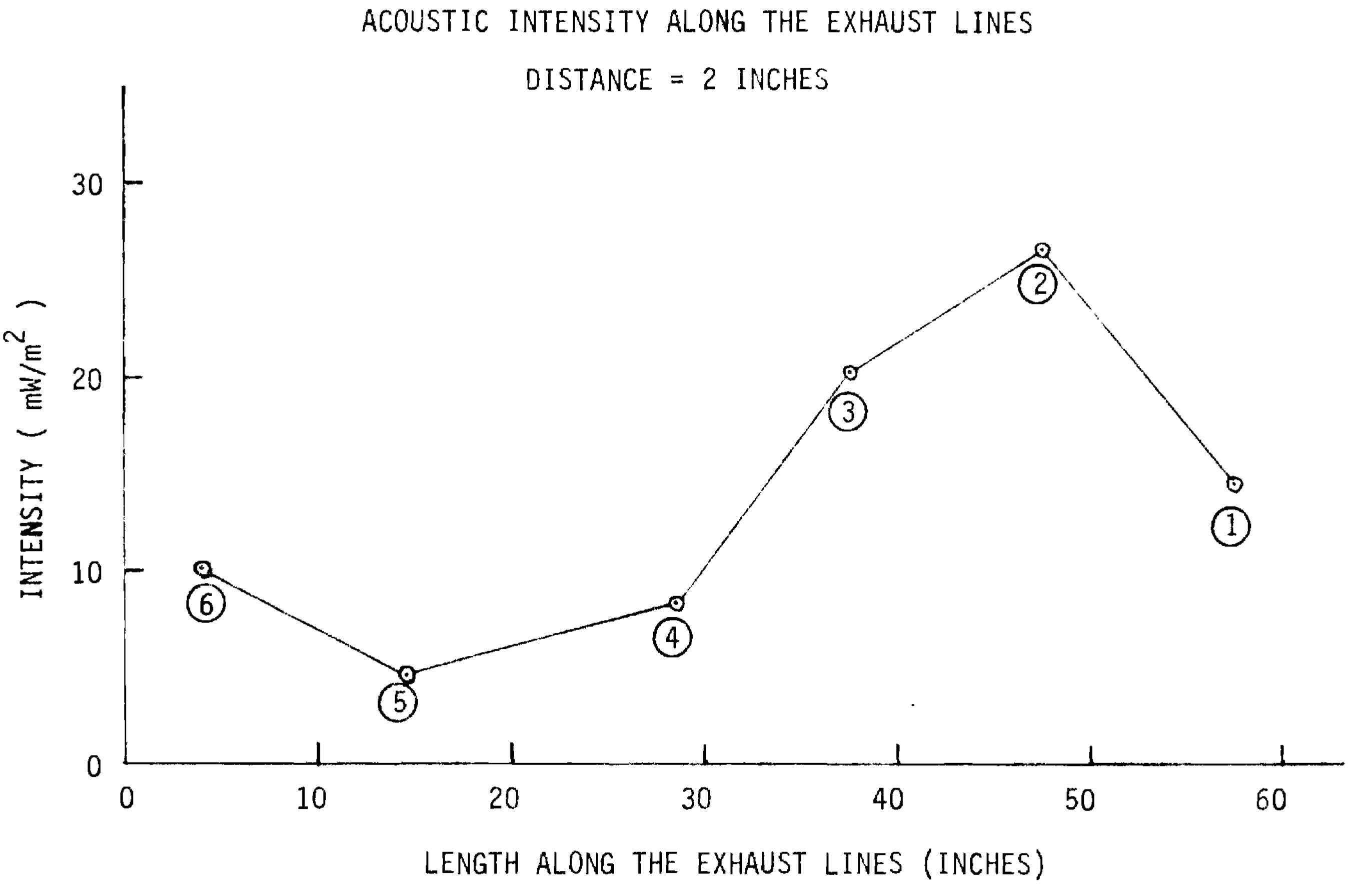


Figure 12. Acoustic intensity survey along the exhaust line

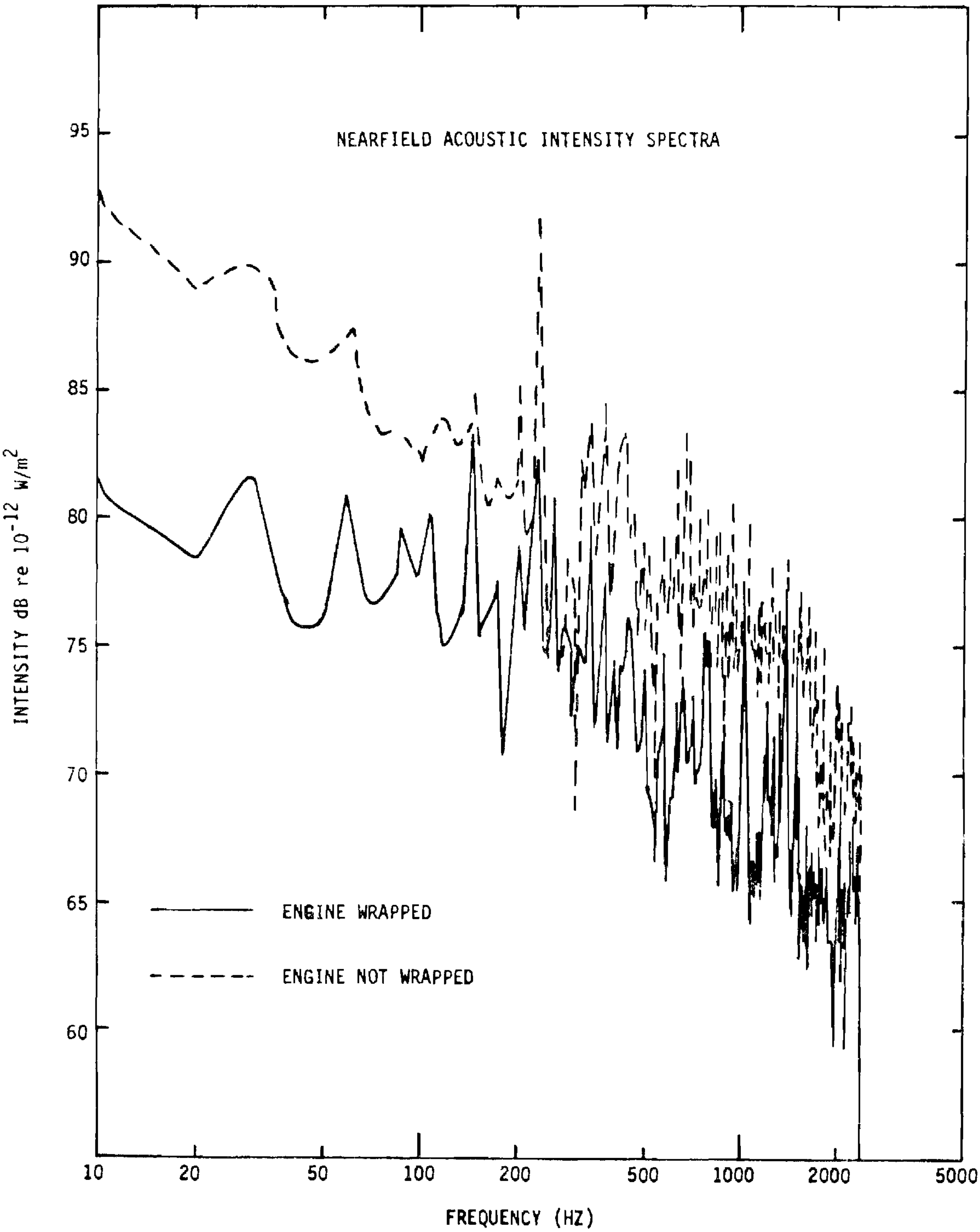


Figure 13. Typical intensity spectra near the turbine for a wrapped and unwrapped engine

COHERENCE BETWEEN TURBINE INLET PRESSURE AND ACOUSTIC PRESSURE

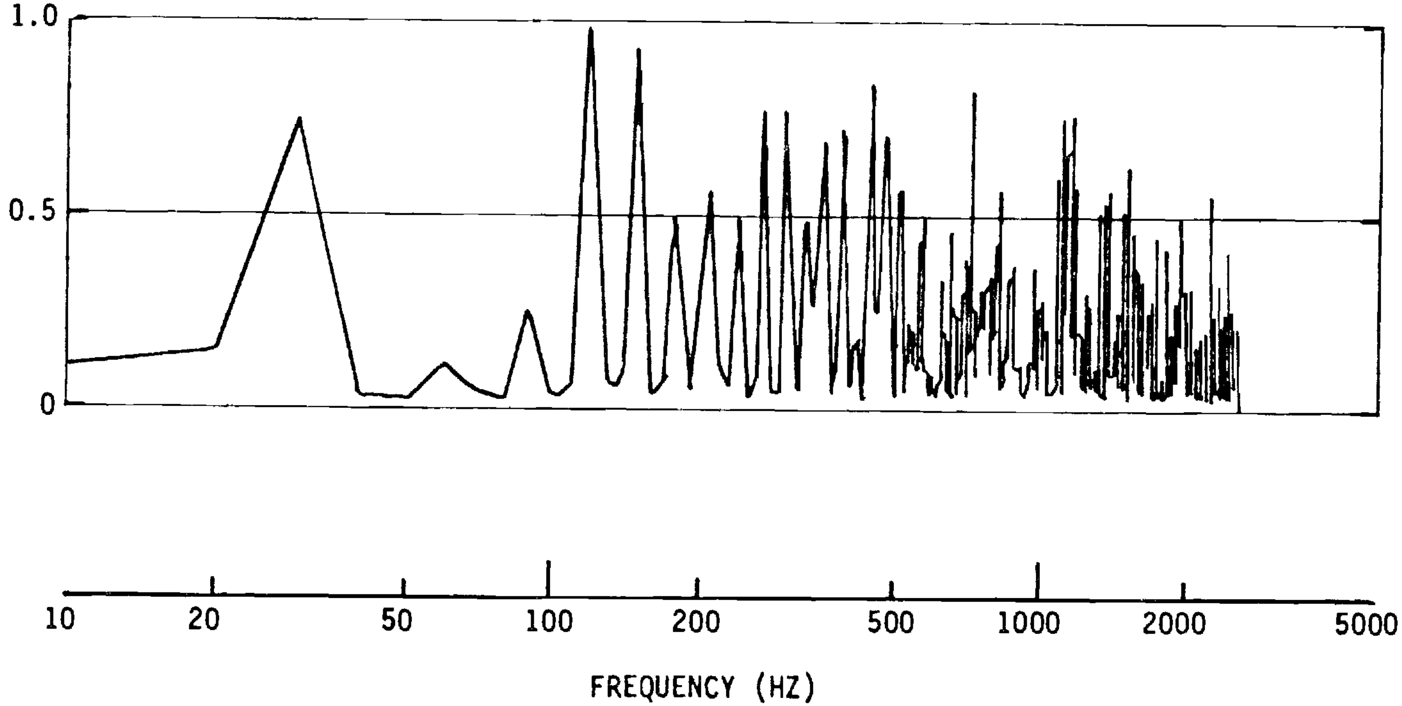


Figure 14. Coherence between turbine inlet pressure and the acoustic pressure 2 inches away from the turbine

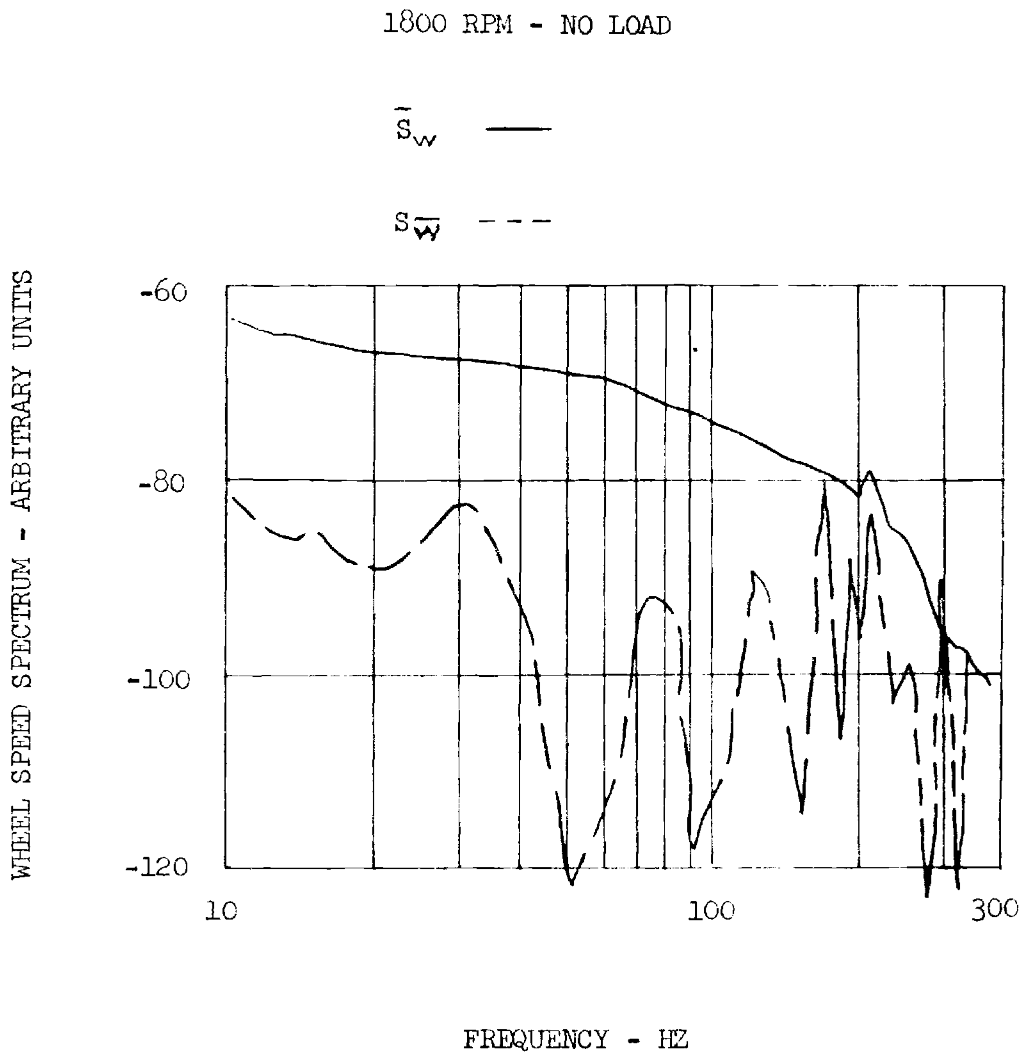


Figure 15. Ensemble averaged spectrum of the wheel speed and the spectrum of the ensemble average

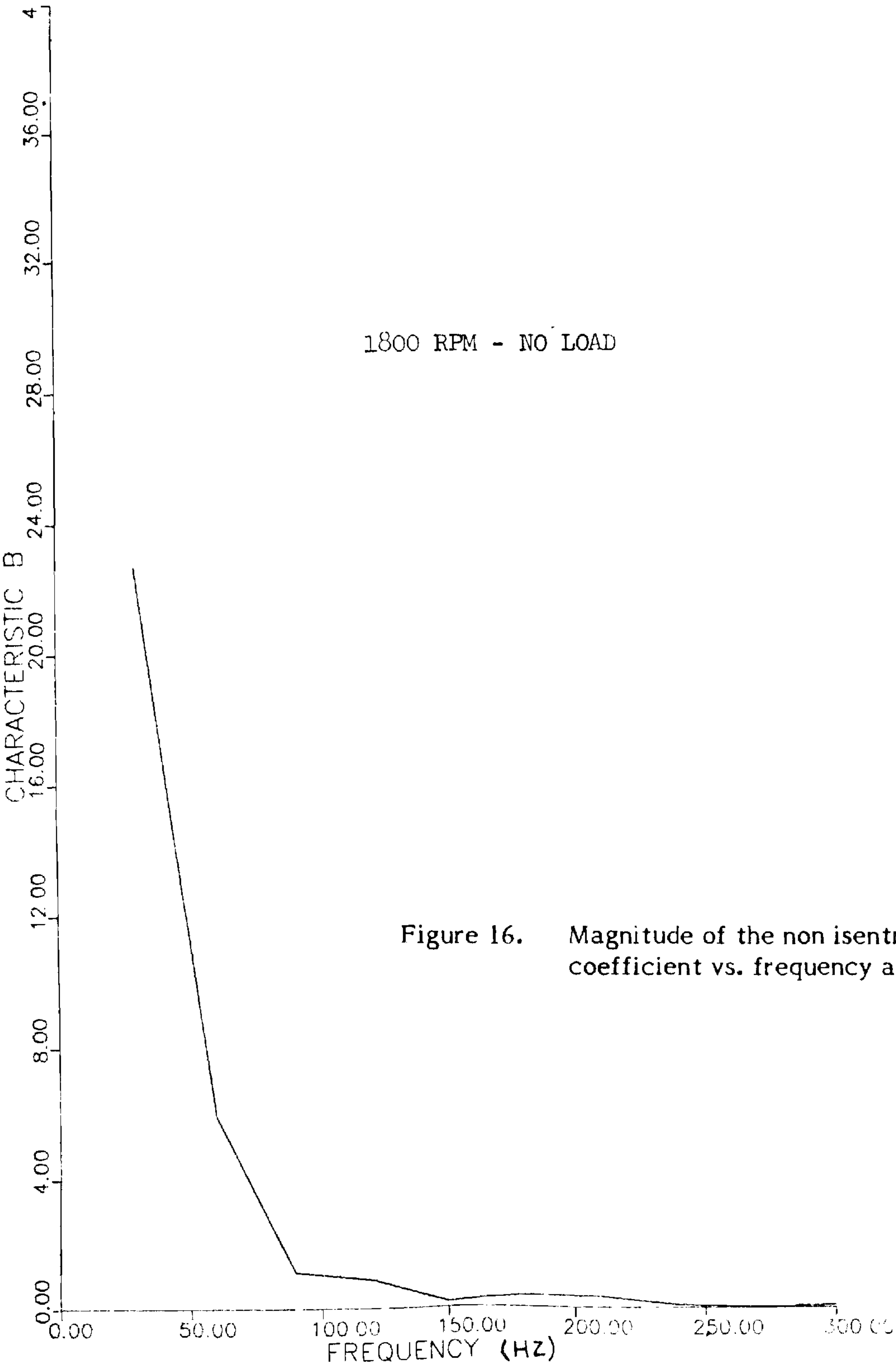


Figure 16. Magnitude of the non isentropic admittance coefficient vs. frequency at no load, 1800 RPM

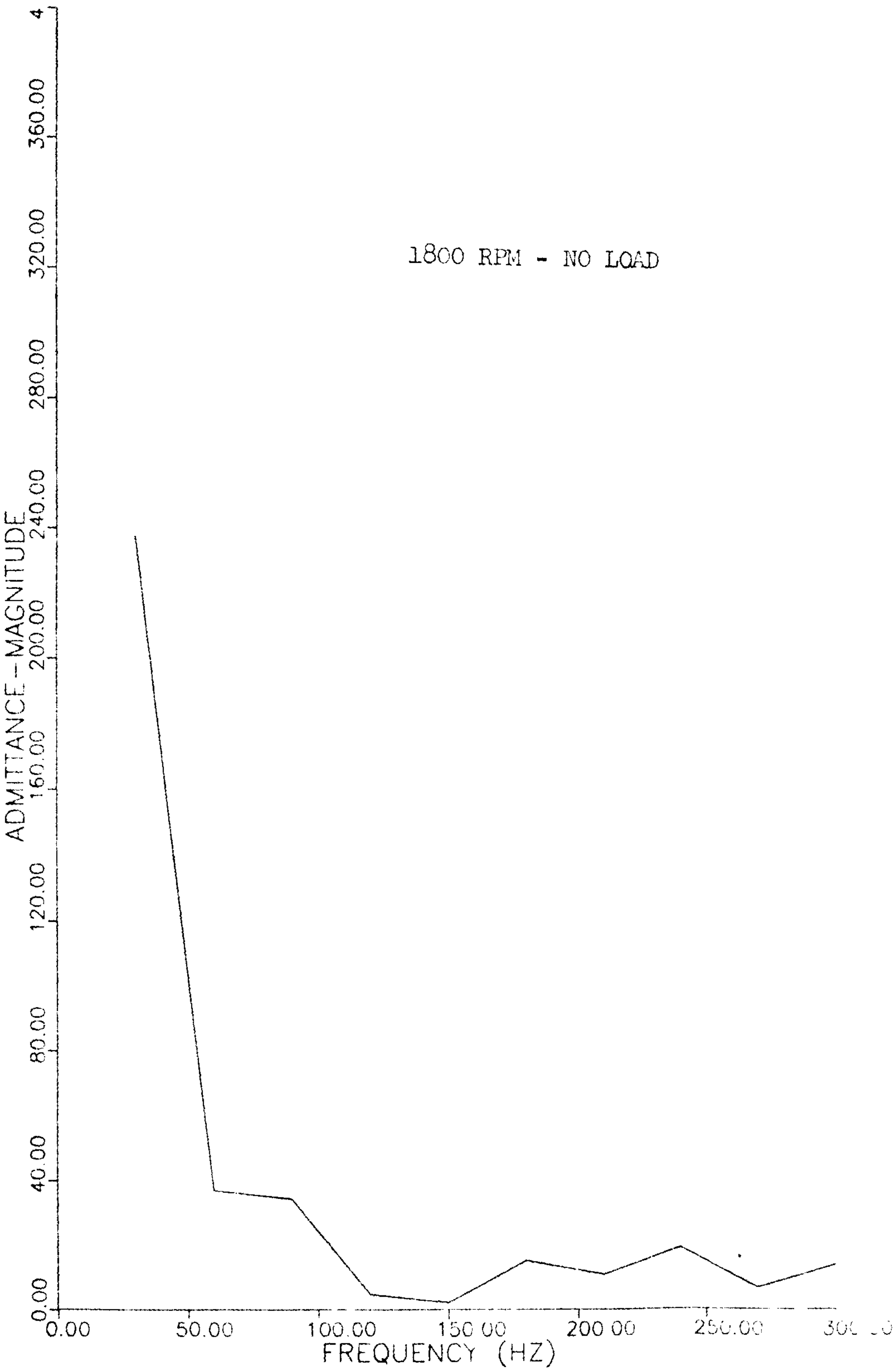


Figure 17. Magnitude of the isentropic admittance coefficient vs. frequency at no load, 1800 RPM

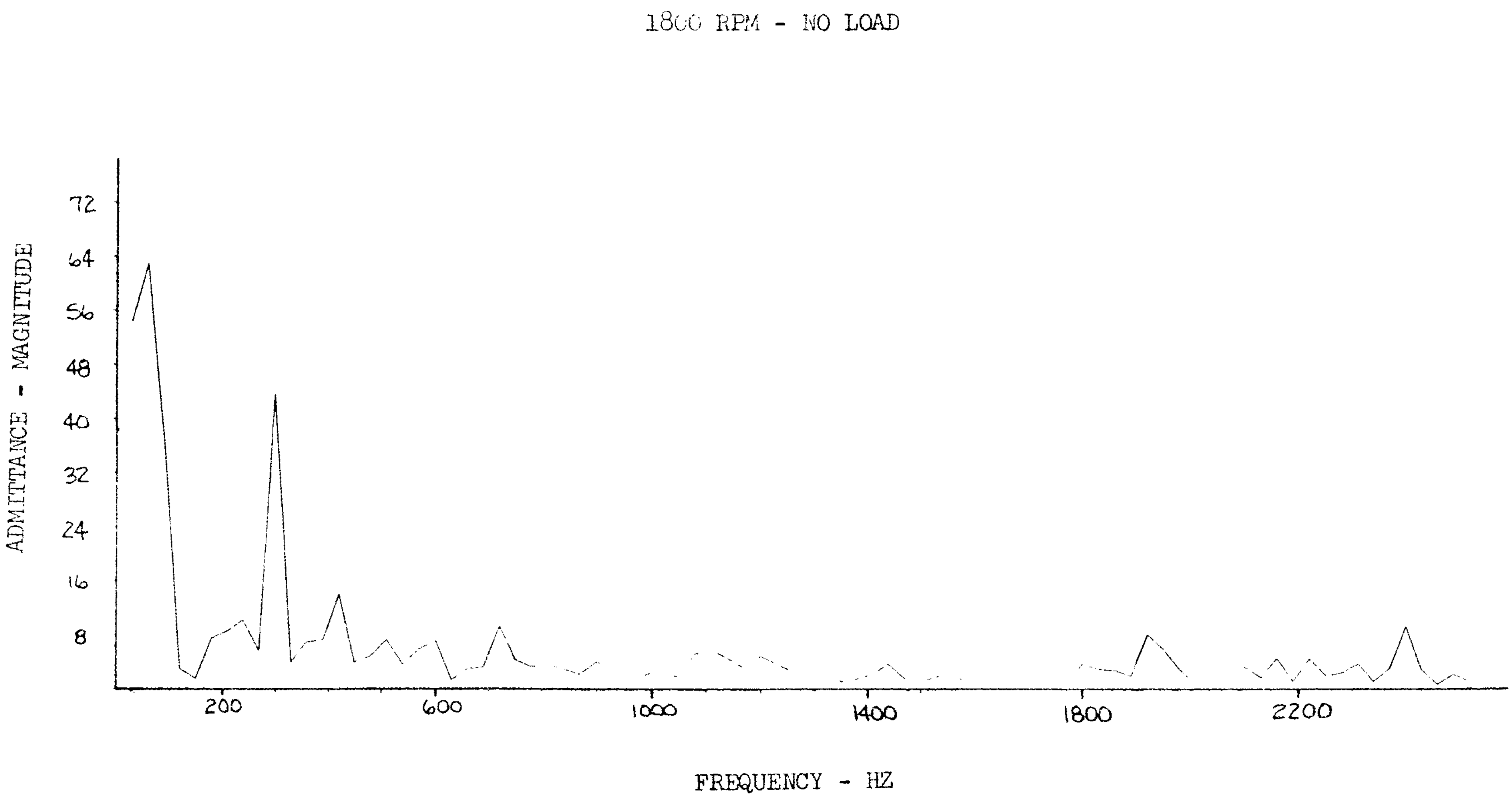


Figure 18. Magnitude of the isentropic admittance coefficient vs. frequency at no load, 1800 RPM assuming B = 0

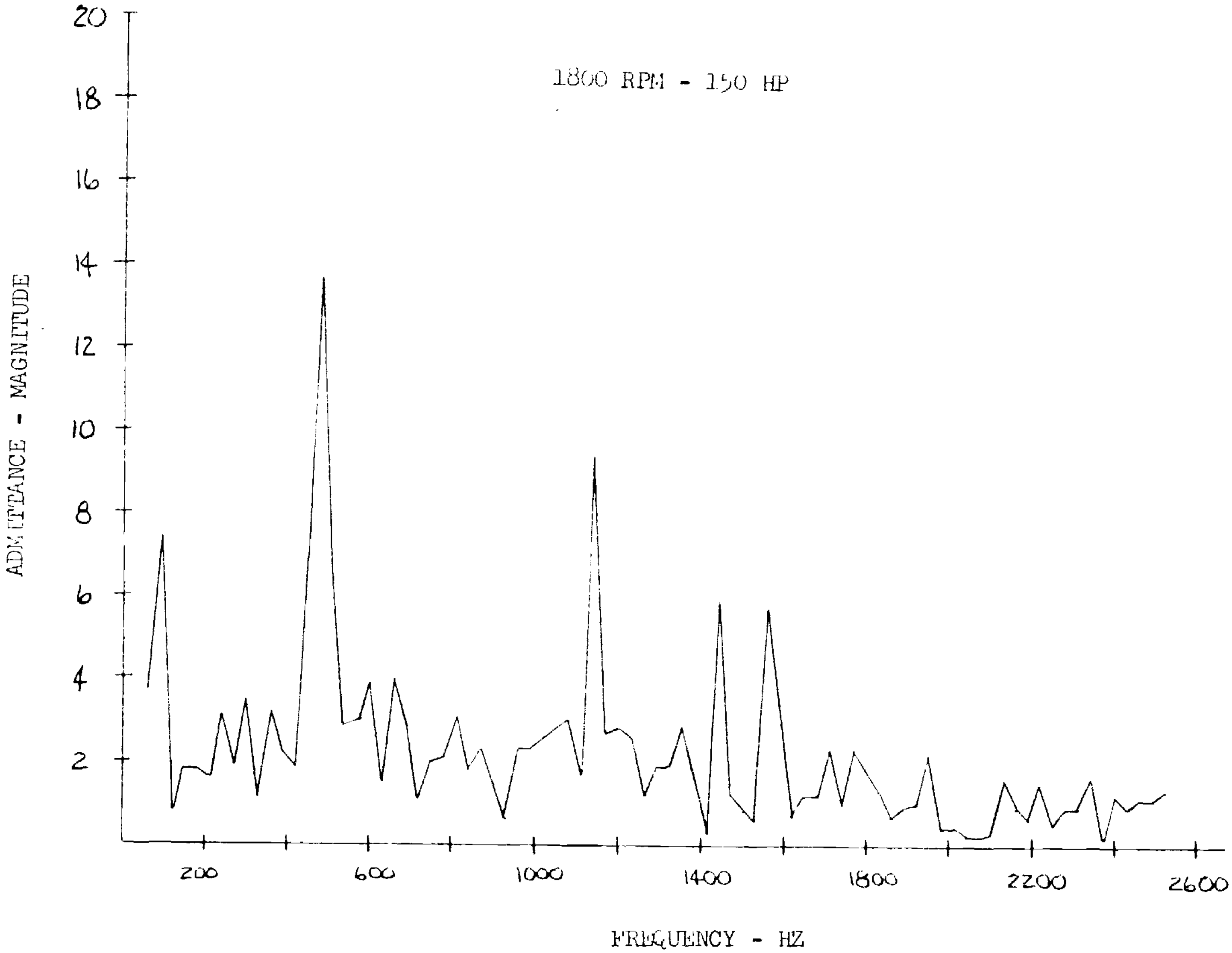


Figure 19. Magnitude of the isentropic admittance coefficient vs. frequency at 150 HP, 1800 RPM